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JOURNAL OF THE BALKAN TRIBOLOGICAL ASSOCIATION

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Journal of the Balkan Tribological Association is an International Journal edited by the Balkan Tribological Association for rapid scientific and other information, covering all aspects of the processes included in overall tribology, tribomechanics, tribochemistry and tribology.
The Journal is referring in Chem. Abstr. and RJCH (Russia).

Aims and Scope

The decision for editing and printing of the current journal was taken on Balkantrib'93, Sofia, October, 1993 during the Round Table discussion of the representatives of the Balkan countries: Bulgaria, Greece, Former Yugoslavian Republic of Macedonia, Romania, Turkey and Yugoslavia. The Journal of the Balkan Tribological Assosiation is dedicated to the fundamental and technological research of the third principle in nature – the contacts.
The journal will act as international focus for contacts between the specialists working in fundamental and practical areas of tribology.
The main topics and examples of the scientific areas of interest to the Journal are:

(a) overall tribology, fundamentals of friction and wear, interdisciplinary aspects of tribology;
(b) tribotechnics and tribomechanics; friction, lubrication, abrasive wear, boundary lubrication, adhesion, cavitation, corrosion, computer simulation, design and calculation of tribosystems, vibration phenomena, mechanical contacts in gaseous, liquid and solid phase, technological tribological processes, coating tribology, nano- and microtribology;
(c) tribochemistry – defects in solid bodies, tribochemical emissions, triboluminescence, tribochemiluminescence, technological tribochemistry; composite materials, polymeric materials in mechanics and tribology; special materials in military and space technologies, kinetics, thermodynamics and mechanism of tribochemical processes;
(d) sealing tribology;
(e) biotribology – biological tribology, tribophysiotherapy, tribological wear, biological tribotechnology, etc.;
(f) lubrication – solid, semi-liquid lubricants, additives for oils and lubricants, surface phenomena, wear in the presence of lubricants; lubricity of fuels;
(g) ecological tribology; the role of tribology in the sustainable development of technology; tribology of manufacturing processes; of machine elements; in transportation engineering;
(h) management and organisation of the production; machinery breakdown; oil monitoring;
(j) European legislation in the field of tribotechnics and lubricating oils; tribotesting and tribosystem monitoring;
(k) educational problems in tribology, lubricating oils and fuels.

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Compiled by S. K. Ivanov, Zh. D. Kalitchin, J. P. Ivanova, M. I. Boneva, N. Evtimova and E. Tosheva.

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Friction

ESTIMATION OF THE STATIC FRICTION COEFFICIENT FOR PRESS FIT JOINTS

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ABSTRACT

The static friction force determines the strength of many different types of joints subjected to large loads, such as press fit joints. In order to calculate the strength, designers need to estimate the real value of the friction coefficient. The new calculation procedure based on the molecular-mechanical theory of friction is proposed in this paper. This procedure considers roughness parameters and hardness of contact surfaces, as well as the relationship between the deformation component of the static friction coefficient and the total static friction coefficient determined experimentally for specific tribological conditions. Studied tribological conditions in the research are related to the press fit joints of railway vehicles drive unit components. The results of the proposed calculation are verified by experimental analysis and in industrial practice.

Keywords: static friction coefficient, press fit joint, real area of contact, surface profile parameters, penetration depth.

AIMS AND BACKGROUND

Most machine designers take design of mechanisms and machines with insufficient concern for friction effects. Designers are acquainted with the fact that friction exists in all sliding systems, but they neglect it. They presume that friction forces are much smaller than forces required for operation function. However, in certain design situations, it is necessary to consider friction forces of devices contact pair for accomplishing successful designs. The static friction force determines the strength of many different joints which transmit large loads. Considering that, engineers need to estimate the real value of the friction coefficient

* For correspondence.

which can be used in joints strength calculation. In most of the cases, the friction coefficient is estimated by experimental ways and the average values for different materials are given in the literature. 'But remember that friction is a system effect and not a material property.'[1]

In the real area of contact between two solid bodies, numerous physical processes are carried out. The difference in hardness of contact surfaces leads to the deformation of the contact area of the softer body because tips of the surface roughness of the harder body penetrate the surface of the softer one. The previous statement suggests that the friction force depends on the mechanical properties of the softer element. This dependence is very complex, because there are possible changes of mechanical properties and mutual transition of material in the process of achieving contact.

Generally, changes of mechanical properties (module of elasticity, hardness) affect the friction coefficient lesser than roughness variations. It can be stated that the increasing roughness of contact surfaces increases the static friction force, as well as the press fit joint strength[2–4]. In addition to mechanical properties of materials, the friction coefficient is affected by the thickness and type of surface film, duration of the contact, mutual dissolving of materials in contact, presence of foreign bodies in the contact area, temperature of the surrounding environment, relative humidity, elasticity, tribosystem, etc. The complexity of friction is very well described by Tabor and Bowden: 'Although friction is simple to measure it is complicated to explain'[5]. Complex phenomena in the process of friction have always been a challenge for tribologists, so there are numerous researches and statements on the nature of the process of friction[6–10].

As the value of the static friction force represents the strength of a press fit joint, tribological parameters have major influence on the press fit joint load capacity[9, 11]. Besides press fit joints, the static friction has an important role in transport means, especially in railway and road vehicles. Regardless of the installed power of the drive units, motion of a vehicle is only possible if there is an adequate static friction between drive wheels and the ground.

The paper[12] reviews the measurement and use of static and kinetic friction coefficients, discusses their usefulness, and describes the sources of frictional resistances in terms of shear localisation. This is particularly true for applications in nano-technology and others that differ from typical laboratory size scales.

The static friction and the induced nonlinear behaviour of mechanisms are analysed in order to solve the positioning problems with extremely high precision[13].

The basic parameters of the friction process are researched in a civil engineering structure with sliding bearings loaded with external forces and simultaneously exposed to the influence of heat, in order to perform an adequate analysis of the static friction coefficient[14].

In addition to the experimental analysis, the value of the friction coefficient can be determined by calculation as well. Most authors set models for calculating the friction coefficient on the basis of theoretical and experimental researches[1,6,7,15–19]. These models can be both used for static and kinetic friction, but they require a lot of experimental data. A new calculation model that is more proper for engineering practice is described in this paper. It is based on the molecular-mechanical theory of friction. The proposed model considers experimental research of tribomechanical pairs at which plastic deformations exist in the real area of contact. The developed calculation procedure is verified in industrial practice of assembling press fit joints of railway vehicles drive units.

PRE-SLIDING MOVEMENT

Researching the static friction, Verhovski[6, 15] introduces the term *pre-sliding movement* as a term denoting the interval of invisible relative sliding of contact surfaces and the value of movement preceding the visible and stable sliding. Therefore, the pre-sliding movement is a period of relative movement characterised by an extensive increase of the reactive force and a small increase of movement.

Press fit joints, screw and rivet connections, all types of friction transmitters (variators, belt transmitters, couplers), parking brakes, etc. work in the mode of pre-sliding movement.

Figure 1 gives a schematic view of a typical press-out process of a shaft-hub system under the act of a longitudinal force. The diagram $F(t)$ shows that force increases from point O to point A, where the maximum value of the press-out force is achieved. That is the static friction force (F_s). After this point, the force rapidly decreases from point A to point B, and in point B it reaches the value of kinetic friction force (F_k). The initial part of diagram OA represents an increase of the force without separation of parts, while the part of the movement diagram $s(t)$ performed in this period presents the pre-sliding movement. The part of diagram AB, where the force decreases from the static force to the kinetic friction force, points to the relative sliding of the components of the press fit joint, which can be seen as a jump in the movement diagram. After point B, the force continuously declines up to point C until the end

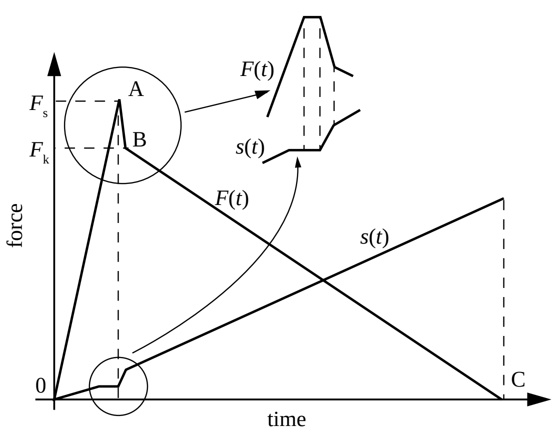


Fig. 1. Schematic presentation of the press-out process

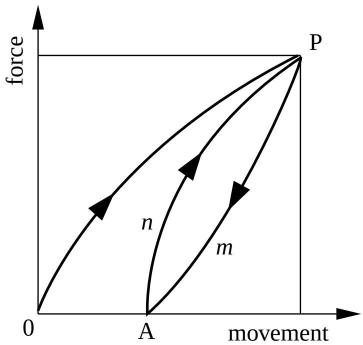


Fig. 2. Hysteresis loop in the pre-sliding phase

of the press-out process, with a relatively stable increase of the movement. Figure 1 also gives an enlarged segment of the process at the time of the sliding start. There one can see that the force retains the value of the static friction force (F_s) for a short time period and then decreases to the value of the kinetic friction force (F_k). This process is followed by an intensive increase of movement.

Verhovski thoroughly researched the phenomenon of the pre-sliding movement up to the start of sliding. He determined that there is a direct proportionality between the tangent force and the movement, but only at the beginning of the pre-sliding movement. Approaching the macro-movement of the frictional joint, the linear dependence between the force and movement disappears because the micro-movement grows faster and the functional dependency between them becomes nonlinear[6, 15]. The hysteresis loop (Fig. 2) is also noticed in the period of the pre-sliding movement. A body, subjected to a tangent force, reaches position P. If then the tangent force is reduced to zero, then the body returns in the reverse direction (*m*). However, the body does not return to the initial position, but stops at some distance OA in the direction of the acting force. That is, the return of the body is not complete. There is a lag in the direction of the tangential force. If the same tangent force acts again, then the body moves (*n*) to the initially achieved position P.

In this way an important effect of the static friction has been determined: there is an accumulation of energy during the pre-sliding movement, but simultaneously with the accumulation there is also some energy dissipation. The dissipation appears in the pre-sliding movement region and it is the main reason why the so-called hysteresis losses exist. The existence of the hysteresis loop shows that the static friction, i.e. the friction in the mode of pre-sliding movement, is a reversible process in some extent.

SOLID CONTACT SURFACE

It is important to understand what a solid surface looks like and how it is deformed when a similar surface is pressed on to it and then slid over it. Bowden said[5]: 'Putting two solids together is rather like turning Switzerland upside down and standing it on Austria. The area of intimate contact will be small.'

Real surfaces of metal parts are rough. Most engineering surfaces are prepared by turning or milling and finished by grinding or polishing. The surface material in the contact area is heavily deformed and as a result the surface layer is often harder than the material below the surface. On top of this deformed material, if the metals are reactive, a layer of metal oxide is formed and this is covered with absorbed molecules of oxygen and water vapour.

The mostly used method of measuring surface roughness is the stylus method. The basic roughness parameters are determined according to international and national standards such as ISO 4287 and ASME B46.1 and similar. The most common roughness parameters are: arithmetical average deviation of the assessed profile (R_a), maximum profile peak height (R_p), average maximum height of the profile (R_z), maximum roughness depth (R_{max}), and profile bearing ratio (t_p).

According to Kombalov[6, 7, 15], the surface roughness can be very well described with the complex roughness parameter Δ, which is calculated by the following expression:

$$\Delta = \frac{R_{max}}{rb^{1/\nu}} \tag{1}$$

where r is the geometric mean value of roughness tips radius in the transverse (r_p) and lateral direction (r_u):

$$r = \sqrt{r_p r_u}. \tag{2}$$

The values of the constants b and v depend on the distribution of the material in the rough surface layer. They can be determined by recording the surface profile with special tribometric measuring devices. The value of the constant b usually ranges within 2 to 4, while the value of the constant n is between 1.5 and 3.

If the values of the roughness parameters R_{max}, R_a and R_p are known, then the values of the constants b and ν can be calculated using the following expressions:

$$b = t_m \left(\frac{R_{max}}{R_p} \right)^{\nu} \tag{3}$$

$$\nu = 2t_m \frac{R_p}{R_a} - 1 \tag{4}$$

where t_m represents the value of the profile cut-off on the medium line of a profile.

In expressions (3) and (4), the parameter values R_{max}, R_a, R_p and t_m are taken as mean values obtained by measuring on at least 5 sampling lengths of the surface profile.

STATIC FRICTION COEFFICIENT

Friction has been a research subject of many scientists during the last five centuries. The knowledge about friction has been developing since Leonardo da Vinci and its fundamental principles about friction in 1495, followed by Newton, Amonton and Desaguliers, Euler, Coulomb, Tomlinson and many others. The most widely accepted friction theories are the Adhesion theory of friction of Bowden and Teibor and Molecular-mechanical theory of Krageljski. A historical review of development of scientific thought about friction is given in literature[1, 15, 21].

The molecular-mechanical theory of friction has been developed on the basis of topological analysis of contact surfaces and determination of mechanical properties of tribo-pair elements[6, 7, 15]. Tips of the surface roughness of the harder body penetrate the surface of the softer one in real contact conditions of 2 bodies. The intensive shear deformation of asperities of the surface layer of the softer body occurs during the process of relative sliding. The opposing force to that deformation is called the deformation component of the friction force. At the same time a force that opposes sliding occurs as well. This force comes from intermolecular interaction in the area of contact. It is called the molecular component of the friction force. Finally, the total friction force that occurs in the real area of contact of a tribo-mechanical system is the sum of molecular and deformation force components:

$$F_{\mu} = F_{\mathrm{m}} + F_{\mathrm{d}}. \tag{5}$$

The values of friction coefficient highly depend on the type of deformations in the real area of contact and geometric contours of microirregularities. Deformations in the area of real contact are determined by mechanical properties of touching solids, the load and the microtopography of contact surfaces. In the areas of real contact, the following types of deformations may occur: elastic, elastic-plastic and plastic. The most common are elastic-plastic ones. However, in many cases for the purpose of calculation, it can be assumed that only elastic deformations occur in the real area of contact on surfaces with surface roughness $R_{\mathrm{a}} \leq 0.2$ µm at contour pressures $p_{\mathrm{c}} < 100$ daN/cm^2, or only plastic deformations on surfaces with surface roughness $R_{\mathrm{a}} \geq 0.8$ µm at contour pressures $p_{\mathrm{c}} > 100$ daN/cm^2. Interactions of solids during elastic-plastic deformations are not researched sufficiently[6, 15].

Plastic deformations in contact area appear when the mean value of the normal stress reaches the value of yield stress. Then, the value of the relative penetration depth is[6]:

$$\frac{h}{r} = 5.4(1-\nu^2)^2\left(\frac{\mathrm{HB}}{E}\right)^2, \tag{6}$$

where h is the depth of the penetration of asperities of the harder body into the contact layer of the softer body; r – the mean value of tips radius of asperities of the harder body; E – elasticity module of the softer body; ν – the Poisson coefficient of the softer body; HB – hardness of the material of the softer body.

In the case of plastic deformations in the area of contact when all (or nearly all) asperities are in contact, that is, in the case of saturated plastic contact, the friction coefficient is determined by the expression[6, 7, 10]:

$$\mu = \frac{\tau_0}{HB} + \beta + 0.9\Delta^{1/2}\left(\frac{pc}{HB}\right)^{1/2}, \tag{7}$$

where τ_0 and β are experimentally determined frictional constants; p_c – contour pressure obtained in the contact. Equation (7) is valid for the bearing area curve of a surface roughness profile characteriяed by parameters $b = 2$ and $v = 2$ obtained by the most common machining processes.

Considering the fact that the total friction coefficient is the sum of molecular and deformation components, for press fit joints with unsaturated plastic contacts, a simplified formula is used for calculation of the static friction coefficient[6, 10]:

$$\mu = \mu_m + \mu_d = \mu_m + 0.5\Delta^{1/2}\left(\frac{p_c}{HB}\right)^{1/4}. \tag{8}$$

Equation (8) considers the basic parameters of a tribo-pair: constructive characteristics (p_c), the quality of surface machining (Δ) and material properties (HB) for the deformation component and tribological conditions (the type of the lubricant is the most important parameter) for the molecular component of the friction coefficient.

The theoretical determination of the value of the molecular component of the friction coefficient is not successful enough due to a large number of influential factors that are difficult to be determined. That is the reason why the molecular component m_m should be determined experimentally[7].

ESTIMATION OF THE STATIC FRICTION COEFFICIENT FOR PRESS FIT JOINTS

The experimental determination of the molecular component of the friction coefficient needs very complex procedure that requires very specific laboratory equipment which is not widely available, so it is not suitable for engineering practice.

In order to simplify the procedure for the estimation of the static friction coefficient the new calculation procedure is proposed based on the knowledge of interrelation of values of the molecular and deformation component of the static friction coefficient. This interrelation depends on the type of deformations in real contact areas and it is, for assemblies of steel parts, mainly $\mu_d/\mu_m = 0.6–1.5$ (Ref. 7). The deformation component can be calculated based on the knowledge of the mechanical characteristics of the material and roughness parameters[6, 7, 10]. Therefore, the problem of calculating the total static friction coefficient is reduced to the determination of the correlation between the molecular and deformation component of the friction coefficient. In this way, the complex experimental determination of the molecular component is avoided.

Hence, the new proposed calculating procedure requires the measured parameters of roughness and hardness of contact surfaces, as well as the ratio of the molecular and deformational components of the friction coefficient that can be obtained experimentally in real friction conditions[5, 9].

For the contact of steel surfaces with plastic deformations in the real area of contact, which is the case in press fit joints of large load capacities, the total value of the static friction coefficient can be calculated according to the new expression that simplifies expression (8):

$$\mu = k\Delta^{1/2}\left(\frac{p_c}{\mathrm{HB}}\right)^{1/4}, \tag{9}$$

where k is the coefficient that depends on the type of machining, roughness of contact surfaces and correlation of surface hardness of assembled parts, the value of the surface pressure, as well as the lubricant. The value of the coefficient k should be experimentally determined in tribological conditions that correspond to exploitation conditions. The penetration depth of asperities of the harder element into the surface layer of the softer element has the greatest impact on the value of the friction coefficient. For calculating the value of the penetration depth, the following approximate formula[7] is used:

$$h \approx 3.4R_a\left(\frac{p_c}{p_r}\right)^{1/3} \tag{10}$$

where p_r is the real pressure in the contact area. Its maximum value, based on previously exposed statements, is equal to the hardness of the softer element ($p_r \approx \mathrm{HB}$).

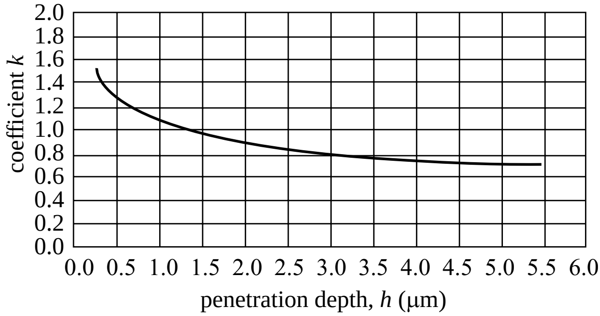
The proposed procedure for determining the static friction coefficient and necessary data are presented in Table 1 (indices o and i denote the outer and inner part, respectively).

During the process of assembling bodies with the curved shape of the body, such as spherical and cylindrical bearings, as well as with flat surfaces of small dimensions, the impact of the specific normal load significantly affects the value of the friction coefficient. This is because the asperities of surfaces are less exhibited and the nominal area of the contact is proportional to the contour area[6]. In that case, for tribological conditions that exist in press fit joints, the contour pressure (p_c) can be considered as the nominal contact pressure.

EXPERIMENTAL VERIFICATION OF THE NEW CALCULATION PROCEDURE

In order to validate the new calculation procedure and to determine the functional dependence between the coefficient k and the depth of the penetration 2 experimental analyses were performed. In the first experiment, the samples with different surface roughnesses were used. Based on the experimental results, the curve

Fig. 3. Dependence of coefficient *k* on the penetration depth *h*



of the coefficient *k* depending on the penetration depth was generated (Fig. 3) (Ref. 9). This dependence is valid for tribological conditions, boundary lubrication by applying the lubricant Loctite Wheel Mount and the contour pressure of about 100 N/mm^2. These tribological conditions correspond with press fit joints of drive unit components of railway vehicles.

In order to validate the new calculation procedure (Table 1) and to determine the static friction coefficient, samples of press fit joints were prepared for the second experiment. The shape and dimensions of a press fit joint sample are shown in Fig. 4. The inner part was made of steel 42CrMoS4 and the outer part of steel C40E. Different surface hardnesses of samples were performed by different heat treatments. The final machining process of the samples was turning or grinding.

Table 1. The proposed procedure for calculating the static friction coefficient

Calculation phase	Neccesary data
Input data	contour pressure: p_c; surface hardness: HB_i, HB_o; roughness of contact surfaces: R_{ai}, R_{maxi}, R_{pi}, t_{mi} R_{ao}, R_{maxo}, R_{po}, t_{mo}
Determining the constant э	$\nu = 2t_m \dfrac{R_p}{R_a} - 1$
Determining the constant *b*	$b = t_m \left(\dfrac{R_{max}}{R_p} \right)^{\nu}$
Determining the complex parameter of roughness Δ	$\Delta = \dfrac{R_{max}}{rb^{1/\nu}}$
Determining the penetration depth *h*	$h \approx 3.4 R_a \left(\dfrac{p_c}{HB} \right)^{1/3}$
Determining the coefficient *k*	the value is experimentally determined
Determining the friction coefficient	$\mu = k\Delta^{1/2} \left(\dfrac{p_c}{HB} \right)^{1/4}$

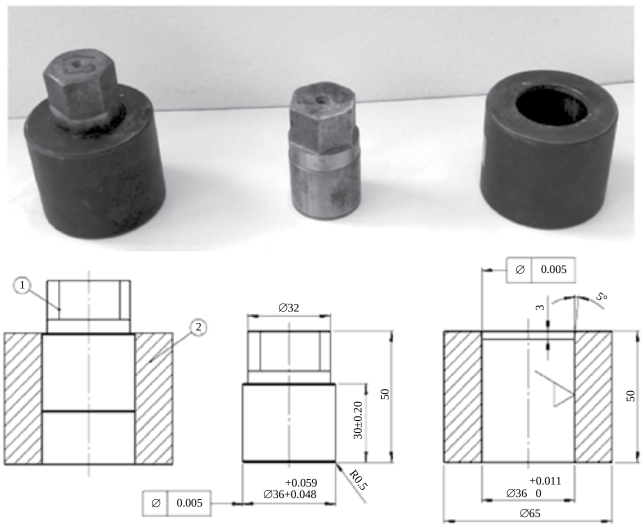


Fig. 4. Press fit joint experimental sample: *1* – shaft; *2* – hub

Before assembling, the contact surfaces of all samples were lubricated by the lubricant Loctite Wheel Mount. The pressing-in process was performed on a hydraulic press.

The proposed calculation procedure was carried out on 21 pairs of experimental samples of shafts and hubs press fit joints. Initially, the surface roughness and hardness of the samples were measured. The surface hardness was measured according to the method of Brinel and the surface roughness was measured by a profilometer. Then, the contour pressure was calculated as contact pressure based on the measured fabrication tightening values and known mechanical properties of materials[20]. The value of the complex roughness parameter D was calculated according to expression (1), based on the measured values of the surface roughness parameters (R_a, R_p, R_{max}, and t_m). After defining all of the necessary parameters, the calculated friction coefficient was determined according to formula (9).

For validation purposes the obtained calculated values of the friction coefficient were compared with the experimentally determined friction coefficient obtained during the pressing-out process of press fit joint samples of shafts and hubs on a hydraulic press. The typical force – movement diagram of the pressing-out process of the experimental sample numbered A17 is shown in Fig. 5.

The experimentally determined friction coefficient represents the ratio of the measured maximum press-out force that represents the static friction force F_s and normal load N, given by expression

$$\mu_s = \frac{F_s}{N} = \frac{F_s}{p_c \pi d l}, \tag{11}$$

where d is the diameter of the press fit joint, and l – the length of the press fit joint.

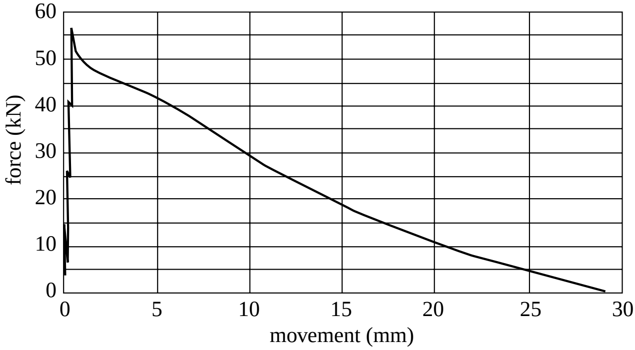


Fig. 5. Force-movement diagram of the pressing-out process of the A17 experimental sample

Data on the tested experimental samples of shaft-hub press fit joints are shown in Table 2.

The analysis of the results in Table 2 shows that despite the similar values of tightening of all press fit joint samples and identical tribological conditions

Table 2. Data on the tested experimental samples of press fit joints

Sample	p_c (N/mm^2)	HB (N/mm^2)	Δ	h (μm)	k	Calculated friction coefficient, μ_c	Experimentally determined friction coefficient, μ_p	Relative error, ε (%)
1	111.02	2099	0.22033	2.350635	1.06	0.2386	0.1934	– 19.0
2	113.89	2129	0.18435	2.115875	1.06	0.2189	0.1747	– 20.2
3	107.38	2129	0.13622	1.86104	0.95	0.1661	0.1802	+ 8.4
4	112.4	2099	0.1419	1.821555	1.045	0.1894	0.1957	+ 3.4
5	103.38	2021	0.14713	1.996134	1.08	0.1970	0.2242	+ 13.8
6	86.35	2021	0.14119	1.891898	1.08	0.1845	0.2258	+ 22.4
7	124.22	2384	0.10669	0.610159	1.175	0.1834	0.1411	– 23.1
8	124.36	2462	0.12626	0.61646	1.17	0.1971	0.1533	– 22.2
9	116.11	2698	0.14281	0.703713	1.16	0.1997	0.1734	– 13.1
10	113.24	2462	0.12863	0.585333	1.173	0.1948	0.1482	– 13.9
11	94.24	2531	0.13467	0.534196	1.185	0.1910	0.1534	– 19.7
12	94.84	2531	0.09603	0.523936	1.19	0.1622	0.1916	+ 18.1
13	90.96	2080	0.111896	1.798673	1.07	0.1637	0.1883	+ 15.0
14	93.97	1913	0.079703	0.722938	1.15	0.1621	0.1960	+ 20.9
15	110.02	1913	0.094529	1.536964	1.083	0.1631	0.1575	– 3.4
16	110.8	2080	0.060198	1.165298	1.115	0.1315	0.1228	– 6.5
17	110.36	1952	0.10994	1.711105	1.08	0.1747	0.1509	– 13.6
18	92.85	1893	0.073281	1.333039	1.13	0.1440	0.1626	+ 12.9
19	97	1834	0.065176	1.277503	1.125	0.1377	0.1462	+ 6.2
20	103.41	1776	0.040256	0.791435	1.14	0.1124	0.1389	+ 23.7
21	94.4	1834	0.102159	1.544516	1.083	0.1649	0.1670	+ 1.4
Average value:						**0.1751**	**0.1707**	**– 0.9**

during the experiment a wide range of experimentally determined static friction coefficients is obtained (from 0.123 to 0.226). It is also observed that, generally, there is a strong correspondence between calculated and experimentally obtained values of static friction coefficients, which indicates that the proposed procedure for calculating the static friction coefficient is reliable enough for using at press fit joints with significant load carrying capacity. Relative errors that occur between calculated and experimentally determined static friction coefficients are in the range of −23 to +24%, which are significantly less than the variations that would be obtained if an unique value of the friction coefficient, according to recommendations from the literature, would be adopted for all experimental samples.

During experimental analysis certain characteristics related to the static friction coefficient were observed, which could be utilised in engineering practice. Especially, it is very important to know the type of the deformation in the real area of contact. If the plastic deformation exists in the real area of contact, the penetration depth represents an important tribological parameter which depends on the roughness parameter and hardness of the contact surfaces. Depending on the value of the penetration depth, the ratio between the deformation component of the static friction coefficient and the total static friction coefficient can be estimated. Based on the experimentally obtained data, a graph of the ratio between the deformation component of the static friction coefficient and the total static friction coefficient in the function of the value of the penetration depth was generated (Fig. 6). It is obvious that increasing the penetration depth increases the share of the deformation component in the total friction coefficient.

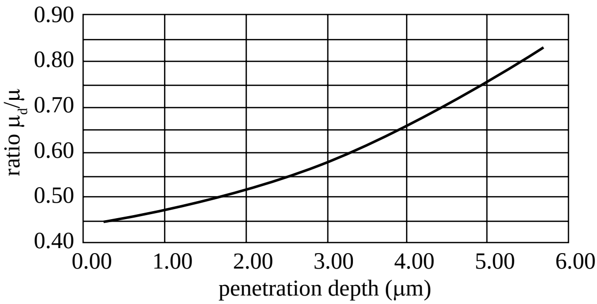


Fig. 6. Ratio between the deformation component and the total static friction coefficient in the function of the penetration depth value

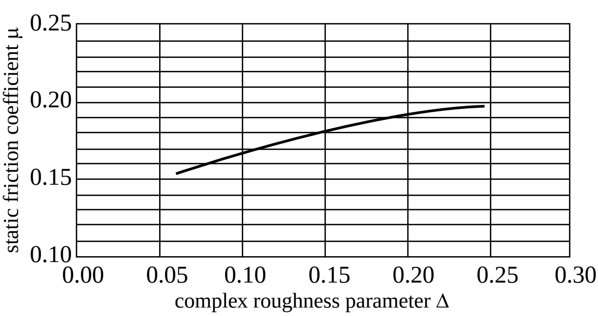


Fig. 7. Dependence of the static friction coefficient μ and the complex roughness parameter Δ at press fit joints

Furthermore, the complex roughness parameter, as a parameter that considers the profile bearing area curve, has a significant impact on the value of the static friction coefficient. Therefore, it is important to determine the dependence of the static friction coefficient and the complex roughness parameter. Figure 7 shows the dependence of the static friction coefficient and the complex roughness parameter at press fit joints, which was obtained in the described research.

CONCLUSIONS

The static friction force determines the strength of many different types of joints which are subjected to great loads, such as press fit joints. The static friction coefficient has large interval of values and depends on many parameters: mechanical characteristics of material, lubricant type, surface roughness and hardness, presence of impurities, contact duration and so forth. The influence of these parameters significantly depends on the type of deformations in the real area of contact.

In order to calculate the strength of a joint, designers need to estimate the real value of the friction coefficient. They usually use average values of the friction coefficient that are estimated experimentally and presented in literature for different materials. In this way, they could make significant errors in design, because friction is not a material property, but a system effect that depends on specific tribological conditions. Numerous authors deal with different approaches in determining the value of the static friction coefficient. But, those approaches require many experimentally data, that is not suitable in engineering practice. In order to simplify the estimation of the static friction coefficient this paper gives a new calculation procedure based on the molecular-mechanical theory of friction. It requires besides measured data of contact surface roughness and hardness only one experiment to be carried out in order to determine the relationship between the deformation component of the static friction coefficient and the total static friction coefficient for specific tribological conditions. This is more convenient for applied engineering.

The proposed calculation procedure was verified by experimental analysis of press fit joints of railway vehicles drive unit components. The presented calculation procedure was implemented for experimental samples of press fit joints in which plastic deformations occurred in the real area of contact. Calculated values of the friction coefficient were compared with the experimentally determined ones. It was concluded that there was a strong correspondence between the calculated and experimentally determined values of the static friction coefficients. The average relative error was around 1%, while the dispersion was in the order of ±20% which is highly acceptable for engineering practice. The proposed calculation procedure was effectively applied in industrial practice for assembling press fit joints of drive unit components of railway vehicles.

ABBREVIATIONS

b – constant
d – diameter of press fit joint (mm)
E – elasticity module (N/mm^2)
F – force (N)
F_d – deformation force component (N)
F_k – kinetic friction force (N)
F_s – static friction force (N)
F_μ – friction force (N)
F_m – molecular force component (N)
h – penetration depth of asperities (µm)
HB – surface hardness
k – experimentally determined coefficient
l – length of press fit joint (mm)
N – normal load (N)
p_c – contour pressure (N/mm^2)
p_r – real pressure in contact area (N/mm^2)
r – geometric mean value of roughness tips radius in the transverse and lateral direction (µm)
R_a – arithmetical average deviation of the assessed profile (µm)
R_{max} – maximum roughness depth (µm)
R_p – maximum profile peak height (µm)
r_p – radius of roughness tips in transverse direction (µm)
R_p – maximum profile peak height (µm)
r_u – radius of roughness tips in lateral direction (µm)
R_z – average maximum height of profile (µm)
s – movement (mm)
t_m – profile cut-off on the medium line of profile (µm)
t_p – profile bearing ratio (µm)
β – experimentally determined frictional constant
Δ – complex roughness parameter
ε – relative error
μ – friction coefficient
μ_c – calculated friction coefficient
μ_d – deformation component of static friction coefficient
μ_m – molecular component of static friction coefficient
μ_p – experimentally obtained friction coefficient
μ_s – static friction coefficient
ν – the Poisson coefficient
τ_0 – experimentally determined frictional constant (N/mm^2)
v – constant.

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